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EFFECT OF SURFACE PROFILES ON CHARACTERISTICS OF CONCENTRIC RECOIL BEARINGS

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An object of the design of a concentric recoil bearing is to insure reliable full film lubrication over as much of the recoil and counter recoil motions as possible. The surface profile has a great effect on the oil film produced in the bearing. A straight bearing profile produces only a thin and unreliable film. A small taper on the leading edge greatly improves the situation. However, a shallow blind pocket in the loaded sector of the bearing was the best surface profile of those studied. Although the pocket was less
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than a mil deep, it resulted in a faster acting bearing and an appreciable thicker oil film. At the highest bearing pressure, the 10 millisecond film thickness was almost twice that of the next best profile studied.

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EFFECT OF SURFACE PROFILES ON CHARACTERISTICS OF CONCENTRIC RECOIL BEARINGS

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FIGURE CAPTIONS

Bearings of the kind used for reciprocating shafts are the common bearings used in guns. These bearings are often highly stressed and the concentric sleeve bearings used in recoil systems of guns in tanks and self-propelled artillery are particularly highly stressed. The gun tube accelerates rapidly from rest reaching about 70 feet per second in just 2 1/2 milliseconds in the case of the M81 152mm gun-launcher used in the Sheridan vehicle. Furthermore, the initial load is very high; it is 144,000 pounds at 1 millisecond for this same cannon. While there has been a great deal of research on bearings for rotating shafts because of their extensive use in industry, there has been very little on bearings for reciprocating shafts which can serve as a guide for the design of recoil bearings. In the full-film lubrication of these bearings, the load is supported by oil entering one side and exiting the other side of the bearing; this is fundamentally different from the situation with a sleeve bearing for a rotating shaft. Of particular importance is the effect of the surface profile on the characteristics of concentric recoil bearings.

BEARING SURFACE PROFILES STUDIED

The information on thrust bearings can serve to a certain extent as a guide for the design of bearings for reciprocating shafts because they have much in common. It must be modified, however, to allow for the different geometry. Instead of a bearing sliding on a flat plane, one has a cylinder sliding within another cylinder.

One surface profile used for thrust bearings of interest in the present case is the tapered land bearing. In a recoil bearing, the

taper should be kept short both to minimize side leakage and because the bearing must operate in the boundary lubrication region during the highly loaded initial portion of the recoil. Another profile of interest is the so-called Rayleigh step. Lord Rayleigh[1] found analytically that the oil film shape which would carry the maximum possible load was step-shaped and so this bearing surface profile was subsequently known as a Rayleigh step. He calculated the optimum proportions for such a profile but he, of course, considered only steady state operation; when the system is at rest and the gun tube begins to move, the situation is very different. It is similar to the "stalled" Rayleigh step considered by Cameron[2] where there is a finite load and zero film thickness. Rayleigh's optimum proportions are not applicable to a recoil bearing because the gun tube rapidly accelerates from rest under very high load and never reaches steady state. Therefore, two Rayleigh step bearings were studied, each with a different step position. With a Rayleigh step used in a sleeve bearing, side leakage could be a problem. To minimize this and because of the extremely thin oil film expected in the initial critical portion of the recoil, the step height was held to only 0.3 to 0.5 mils. A modification of a Rayleigh step which should eliminate excessive side leakage altogether was also studied. This surface profile consisted of a shallow blind pocket where the sides of the pocket minimized the amount of oil leaking from the loaded to the unloaded sectors of the bearing in front of the Rayleigh step.

1Lord Rayleigh, "Notes on the theory of lubrication", Phil. Mag. 35 1-12, (1918)

2A. Cameron, "Principles of Lubrication", p.150-151, John Wiley and Sons Inc., (1966)

The details of the five bearing surface profiles studied are as follows:

1. A straight profile was included to serve as a comparison for the others. Both the front and rear edges were left sharp.

2. The tapered land bearing had a straight profile as above but with a 0.3 to 1.0 mil. taper on its leading edge extending 1/8 in. into it.

3. A Rayleigh step bearing with the step 1/2 in. from the leading edge was studied. The height of the step was 0.3 to 0.5 mils.

4. A Rayleigh step bearing with the step 1 in. from the leading edge was also studied. The height of this step was the same as the above.

5. The blind pocket studied was 1 in. long, 3/8 in. wide, and 0.8 mils deep. It was in the center of the upper (loaded) surface and open at the leading edge.

The dimensions of the test bearings were certainly not optimum but they should allow the characteristics of the various profiles to be compared. The uncertainty in the step and taper heights was because of wear during the experiments. The larger value was the initial measurement, and the smaller, the final.

RECOIL SIMULATOR

A concentric recoil system consists essentially of two sleeve bearings with an oil reservoir between them and oil seals outboard of the bearings. The design of the test device duplicating this arrangement is shown in Fig. 1. The replaceable front and rear bearings were clamped between the center section and the end caps by means of four threaded rods. Easily obtained lip oil seals were mounted in the end caps. This kind of an oil seal is not designed to resist appreciable

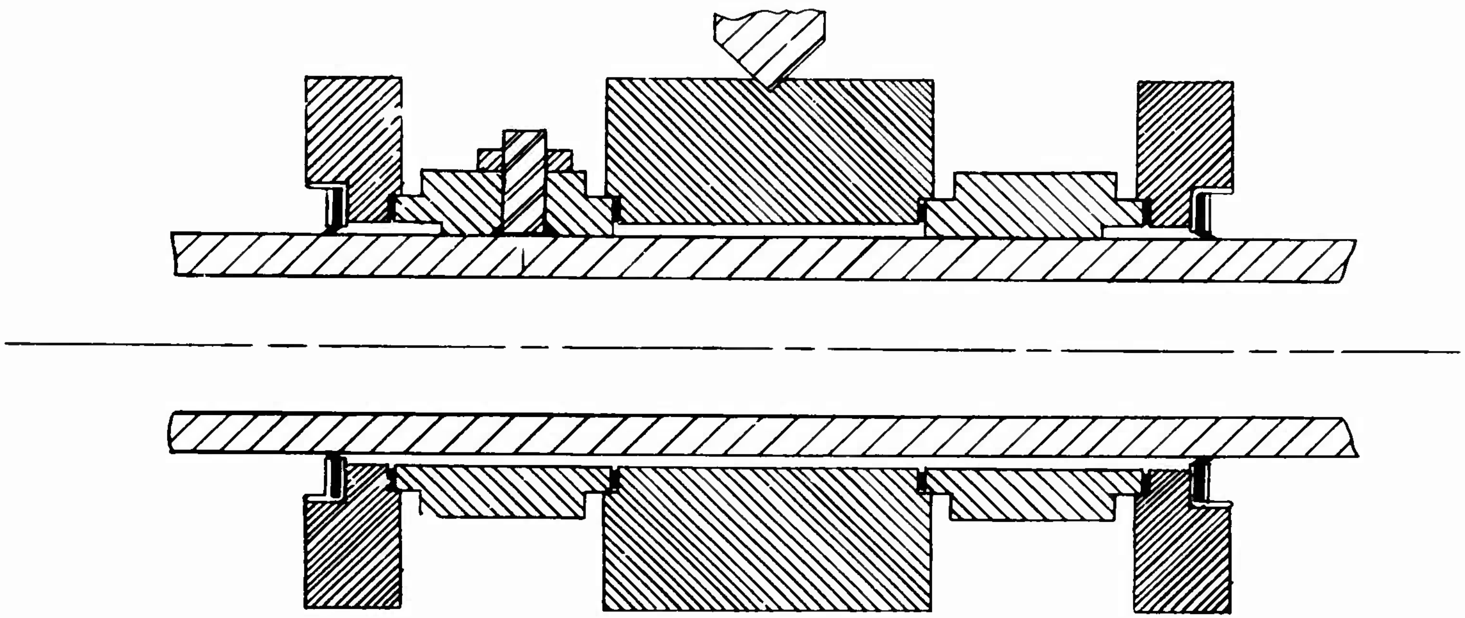

Fig. 1 Cross section of the concentric recoil bearing test assembly.

pressure but since the recoil was so rapid and the oil pressure probably not great because of the design of the system, the leakage was negligible. The center section served as a reservoir for the recoil oil; air was eliminated by introducing the oil under a gravity head at the bottom and venting it at the top.

The gun tube was simulated by a hollow, case-hardened shaft. The shaft was hollow to reduce its inertia so that it could be accelerated as rapidly as possible. It was supported and allowed to move freely back and forth by two linear ball bearing units, one on each end of the recoil-bearing test assembly.

The basic design of the test bearings is shown in Fig. 2. The internal diameter was 2.0005 in., - 0.0000, + 0.0010; the diameter of the simulated gun tube was 1.998 in. so the diametral clearance was 0.0025 to 0.0035 in. The bearing length was 1 5/8 in.; therefore, the projected area was 3.250 sq. in. Originally, the test bearings had been designed 2 in. long but part of the bearing surface was removed in order to obtain higher bearing pressures. The test bearings were all identical with the exception of their surface profiles. A large "U" shaped groove ran the length of the bearing on the bottom (unloaded) surface. The oil which must flow through the bearing at half the relative velocity for full-film lubrication returns to the reservoir through this groove in the case of the rear bearing. The groove in the front bearing allows the oil to reach the leading edge during recoil. Because of the groove, oil is not forced past the rear oil seal nor does the front bearing "starve". There was also a large annular space between the bearings and the oil seals so that the oil could circulate

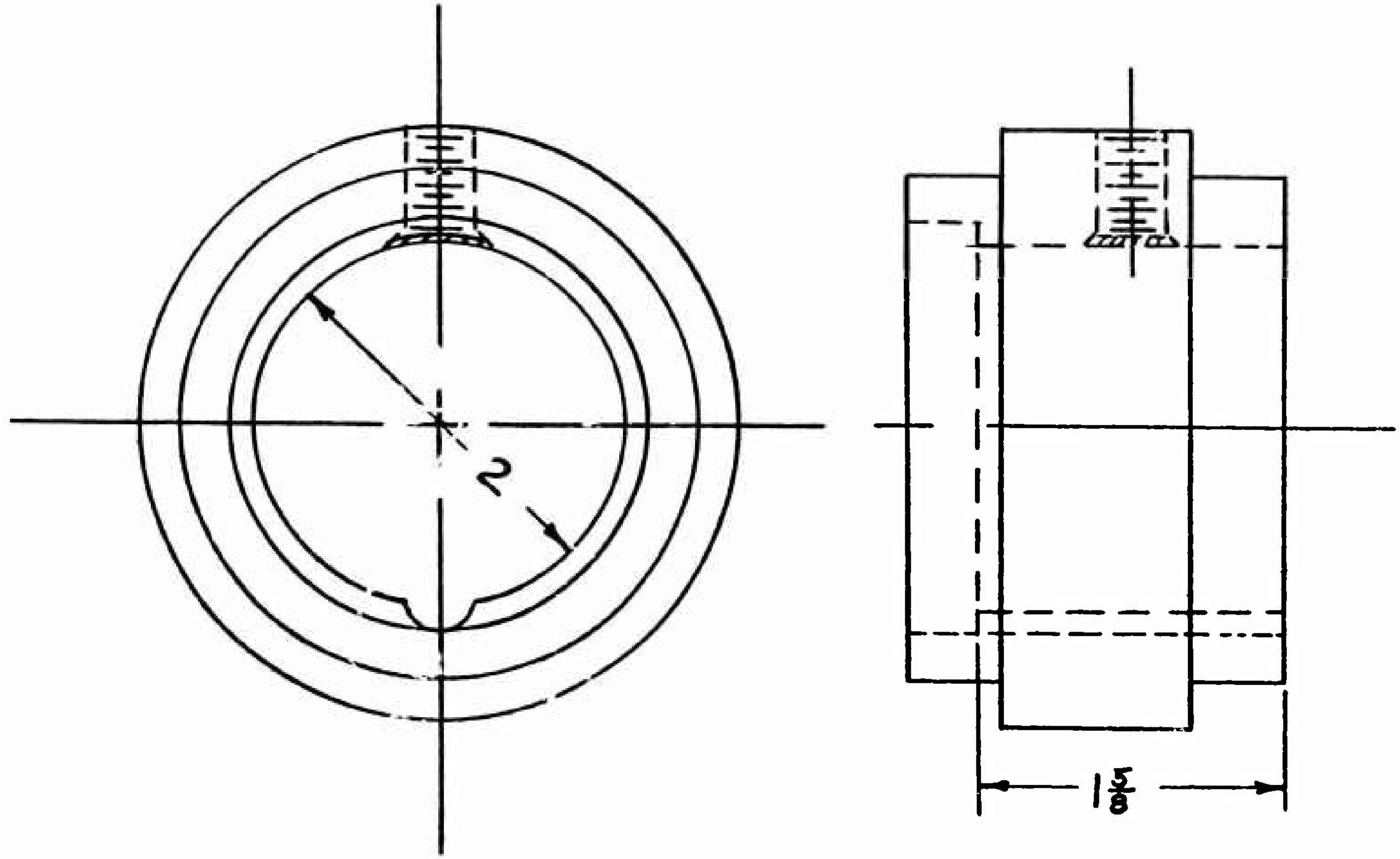

Fig. 2 Basic design of the test bearings.

freely around the simulated gun tube. A Bently Nevada probe and 3000 series proximitor were used to measure the oil film thickness during the experiment. The probe employs magnetic eddy currents to sense the distance to the steel shaft from the probe tip and was positioned on the upper (loaded) surface of the bearing about half way along its length. Metal had to be removed from around the probe tip by counter boring the bottom of the hole so that it would not interfere with the operation of the probe. Furthermore, a thin epoxy window was constructed beneath the probe tip; this window eliminated irregularities on the bearing surface in the high pressure region but was transparent to the magnetic signal. Only the front bearing was used for film-thickness measurements. The probe was calibrated by measuring the signal while rotating the test bearing assembly 180° and comparing it with the exact measured diametral clearance at that point on the simulated gun tube.

For valid measurements, the test device must simulate the important characteristics of a concentric recoil system as closely as possible. In an actual recoil system, both the load and the sliding velocity constantly change throughout the recoil. The gun tube rapidly accelerates and then more slowly decelerates for the remainder of the recoil. (See Fig. 3) The load is initially very high but it falls to a more moderate value in a few milliseconds. (See Fig. 4) Both the load and the sliding velocity on counterrecoil are very much lower. Therefore, the lubrication during the first few milliseconds of recoil is, by far, the most critical. It was not felt that a rapidly changing load would have much effect on the comparison of the different bearing

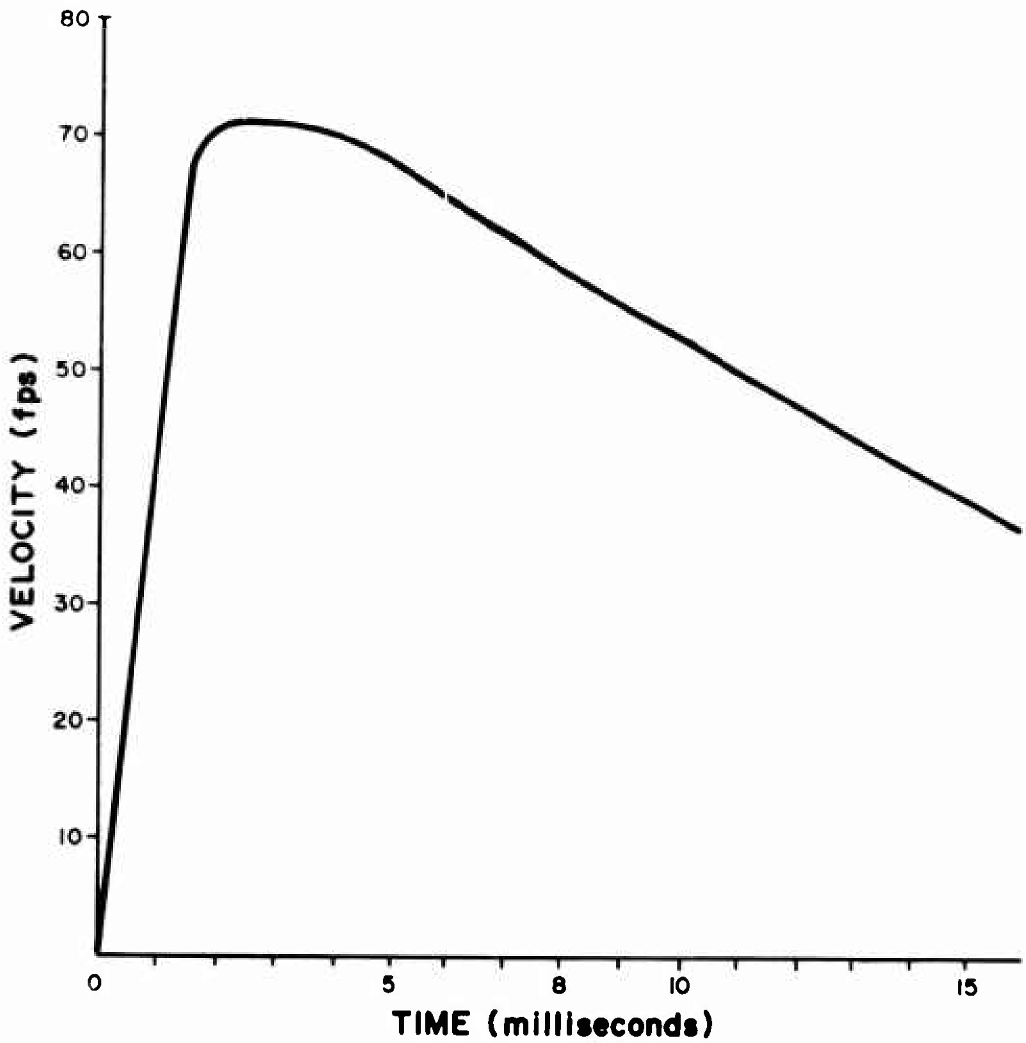

Fig. 3 Velocity as a function of time for the first 15 ms of recoil for the M81 152mm gun-launcher. - The acceleration is not always this rapid. (From data on round 257, UPG Report 0018 by G. S. Groak, Mar. 1970).

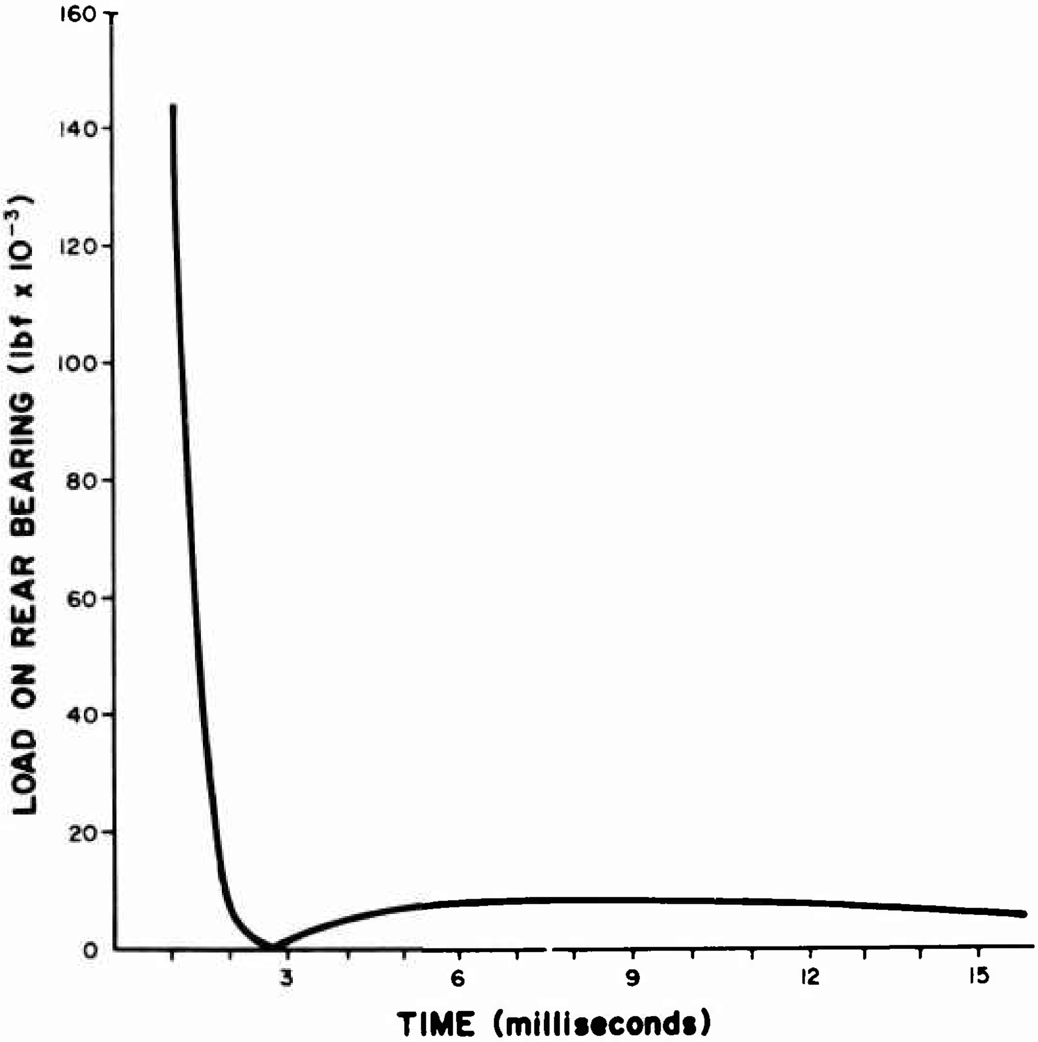

Fig. 4 Load on the rear bearing as a function of time for the first 15 ms of recoil for the M81 152mm gun-launcher.- In the first 2 3/4 ms the loading is downward; after this time, it is upward. (Calculated from the data in Fig. 3).

surface profiles so constant loads were applied while the simulated gun tube was rapidly accelerated. The loads were applied to the assembly by means of a pair of pneumatic power cylinders with a "nut-cracker" arrangement. As much as 1100 lbf could be applied without changing the 250 psi pneumatic regulator or shortening the test bearings. The load system was calibrated by means of a load cell between the load arm and the test assembly.

Rapid acceleration of the simulated gun tube was obtained by striking it with a heavy steel pendulum. While the extremely high accelerations and velocities of an actual gun tube during recoil could not be matched in this way, it was felt that the simulation would be adequate for a comparison of the different test bearings. The simulated gun tube was brought to rest by use of a pneumatic brake. The recoil velocity was measured during recoil by means of a linear-variable-differential-transformer with the core mounted on the end of the simulated gun tube and the body rigidly mounted on the pneumatic brake assembly. The signal from this device as well as the signal from the magnetic oil thickness probe in the test bearing were recorded throughout the recoil using a fast-writing oscillograph. The velocity probe was calibrated by measuring the time and distance of recoil since the time-velocity curve was essentially a triangle when the bearing load was low. A general view of the recoil simulator is shown in Fig. 5.

RESULTS

The five surface profiles were studied at three different bearing pressures, 476, 772, and 1055 psi, with a single hydraulic oil of 103 centistokes viscosity at ambient temperature. Film thicknesses at

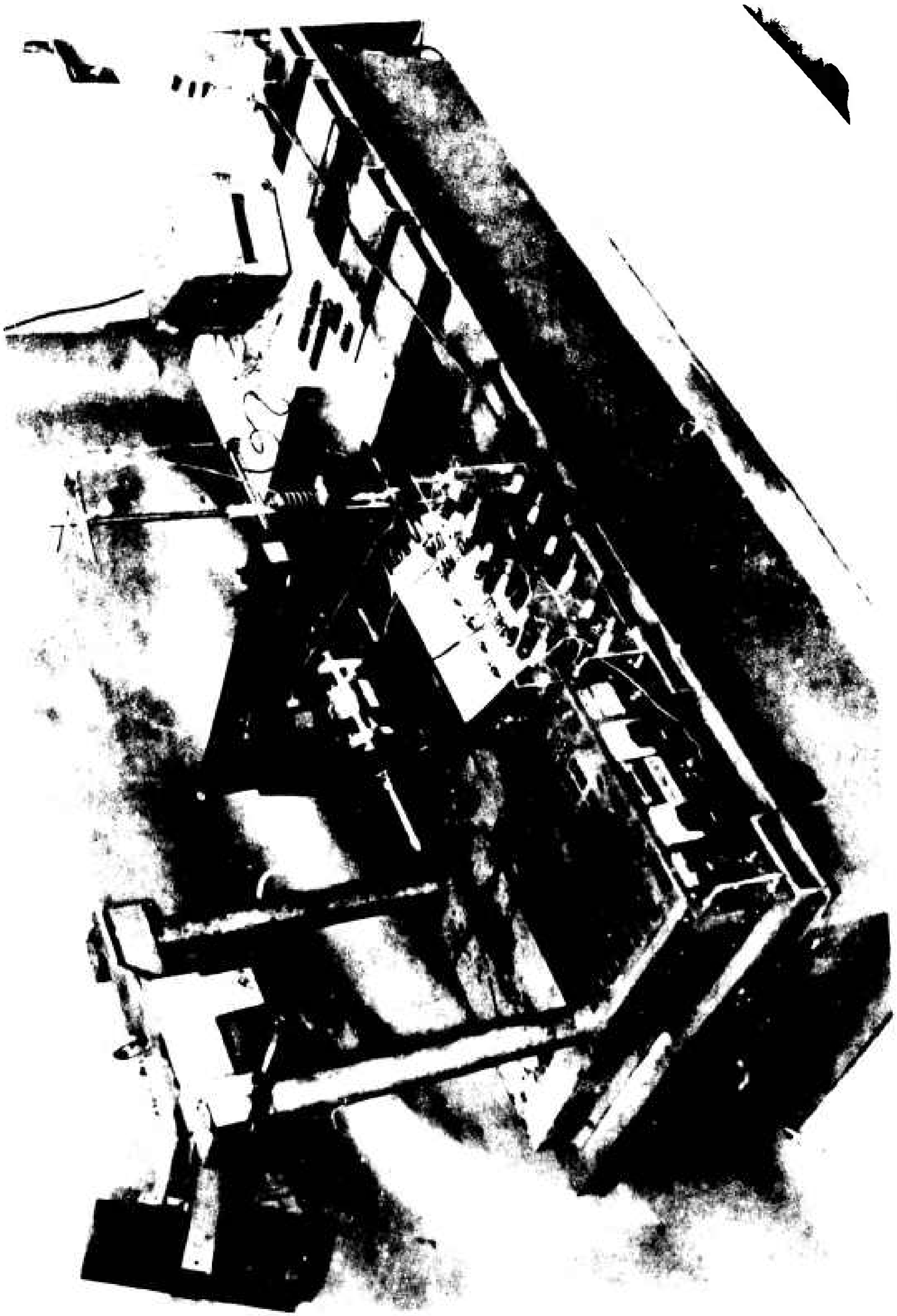

Fig. 5 General view of the recoil simulator.

5, 8, and 10 milliseconds were read from the individual oscillograph records and the arithmetic averages and standard deviations calculated. The velocity record provided a zero-time index. Correlation with run number was tested for every series but none was found and, before data from two series at the same experimental conditions were combined, they were tested statistically to be certain that there was no significant difference. A few measurements which were obviously different from the others were discarded. Also, a few measurements at 8 ms and more at 10 ms were discarded if it appeared likely that the zero of the magnetic probe had changed during the initial portion of the recoil. Data beyond 10 ms were frequently unreliable because of a significant change of zero perhaps related to the heavy impact of the steel pendulum on the simulated gun tube. The number of measurements at each bearing pressure varied from 54 to 117 with most being around 75. There was considerable scatter of the measurements, particularly at the longer times, but the average values fell on smooth film thickness-time curves. The average film thicknesses for the different profiles are plotted as functions of time for 476, 772, and 1055 psi bearing pressures in Figures 6, 7, and 8 respectively and the data are tabulated in Table I.

Straight bearing surface profile

With this profile, a significant oil film was produced in the bearing only at the lowest bearing pressure and, even at this pressure, it was thin and its formation unreliable. The presence of a sharp leading edge is a real danger in practice unless a definite taper, step, etc. is specified. Even if the machinist "broke" the edge, it

would again become sharp as the bearing wore.

Taper land profile

The presence of a taper on the leading edge improved the performance of the bearing immensely. Even with a taper of only 0.3 to 1.0 mils, an oil film was reliably formed and, at the higher bearing pressures, was an order of magnitude thicker than the film for a straight bearing surface profile. Apparently, a large taper is not required for good performance although it must be large enough so that it is not worn away in service leaving a straight surface profile.

Rayleigh steps

The most striking characteristic of both Rayleigh step bearings was that the oil film collapsed rapidly probably owing to excessive side leakage in this design. When the simulated gun tube began decelerating, the film thickness immediately began to decrease. This could mean that the oil film would collapse at the end of the recoil motion so there would be metal-to-metal contact and wear at the beginning of the counter-recoil motion. It could also mean that the oil film would become too thin to maintain full-film lubrication to the end of counterrecoil and so result in "hanging-out-of-battery".

There was considerable difference in the performance of the two Rayleigh step bearings. The thickness of the oil film for the bearing with the step one inch from the leading edge was from one-and-a-half to five-and-a-half times that for the bearing with the step a half inch from the leading edge. The longer length of increased clearance in front of the step did not seem to result in the expected increase in side leakage. The fact that the bearing with the step a half inch

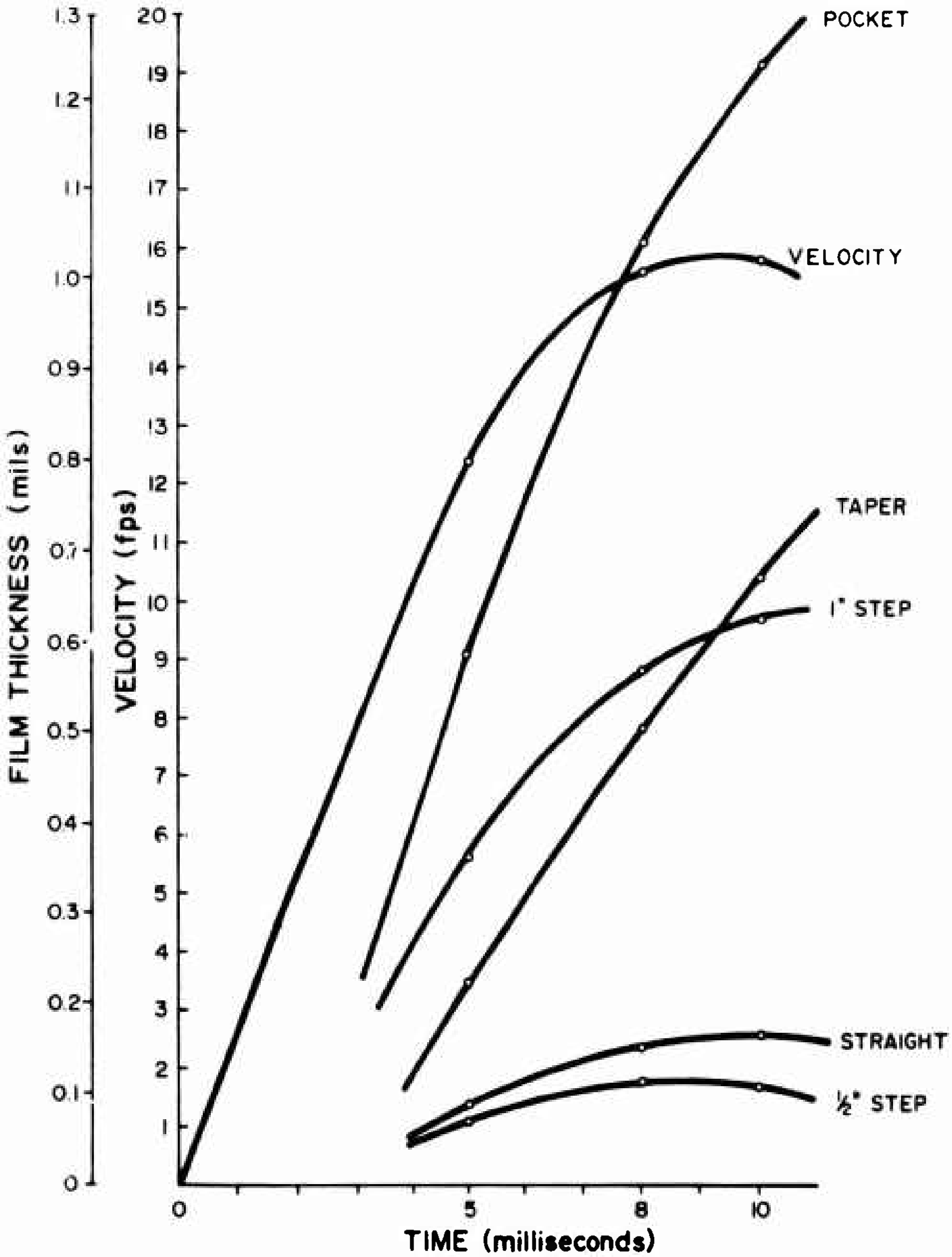

Fig. 6 Average oil film thicknesses for the various bearing surface profiles as functions of time for 476 psi bearing pressure. The test recoil velocity at this particular bearing pressure is included on the same graph.

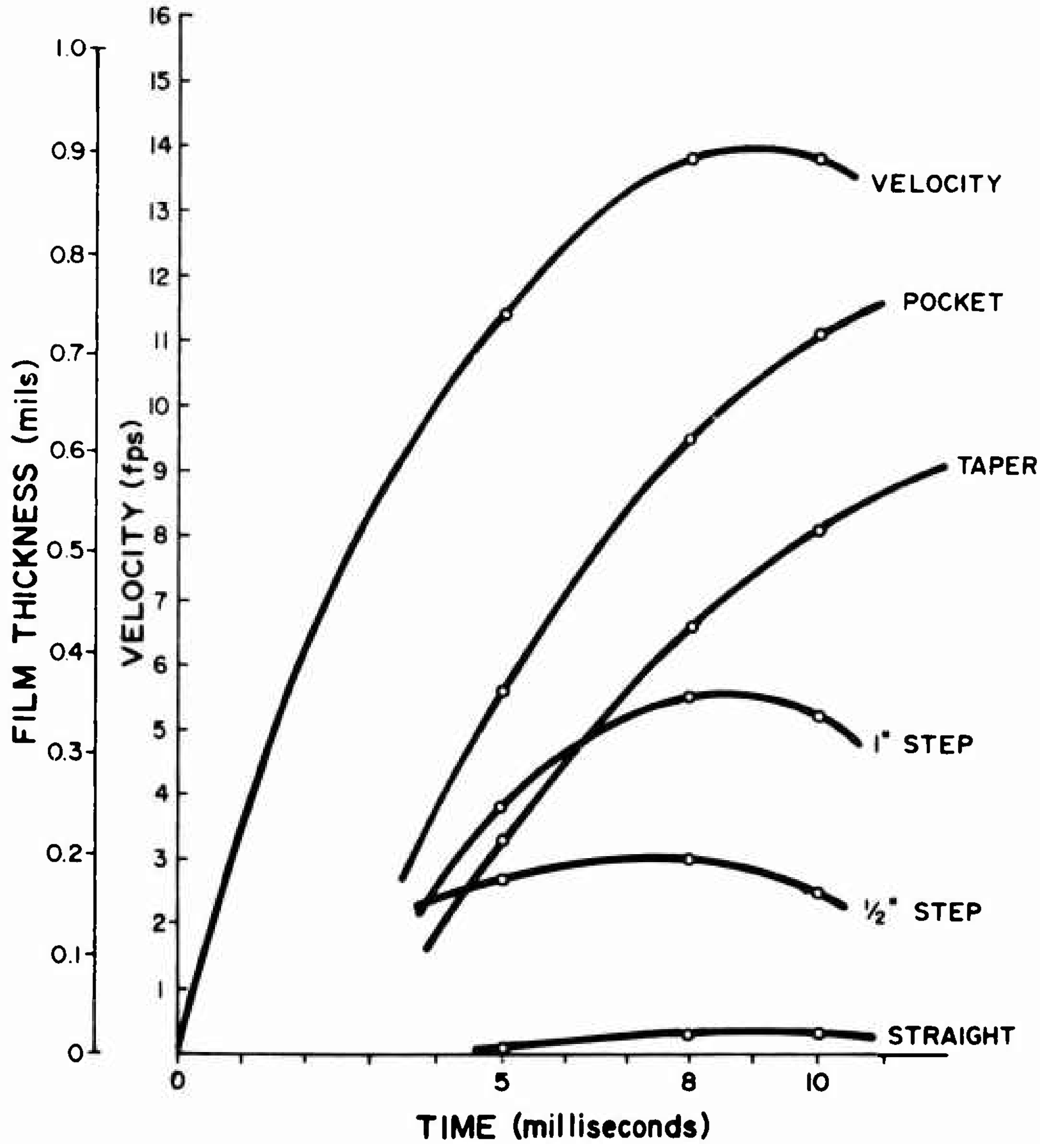

Fig. 7 Average oil film thicknesses for the various bearing surface profiles as functions of time for 772 psi bearing pressure. The test recoil velocity at this particular bearing pressure is included on the same graph.

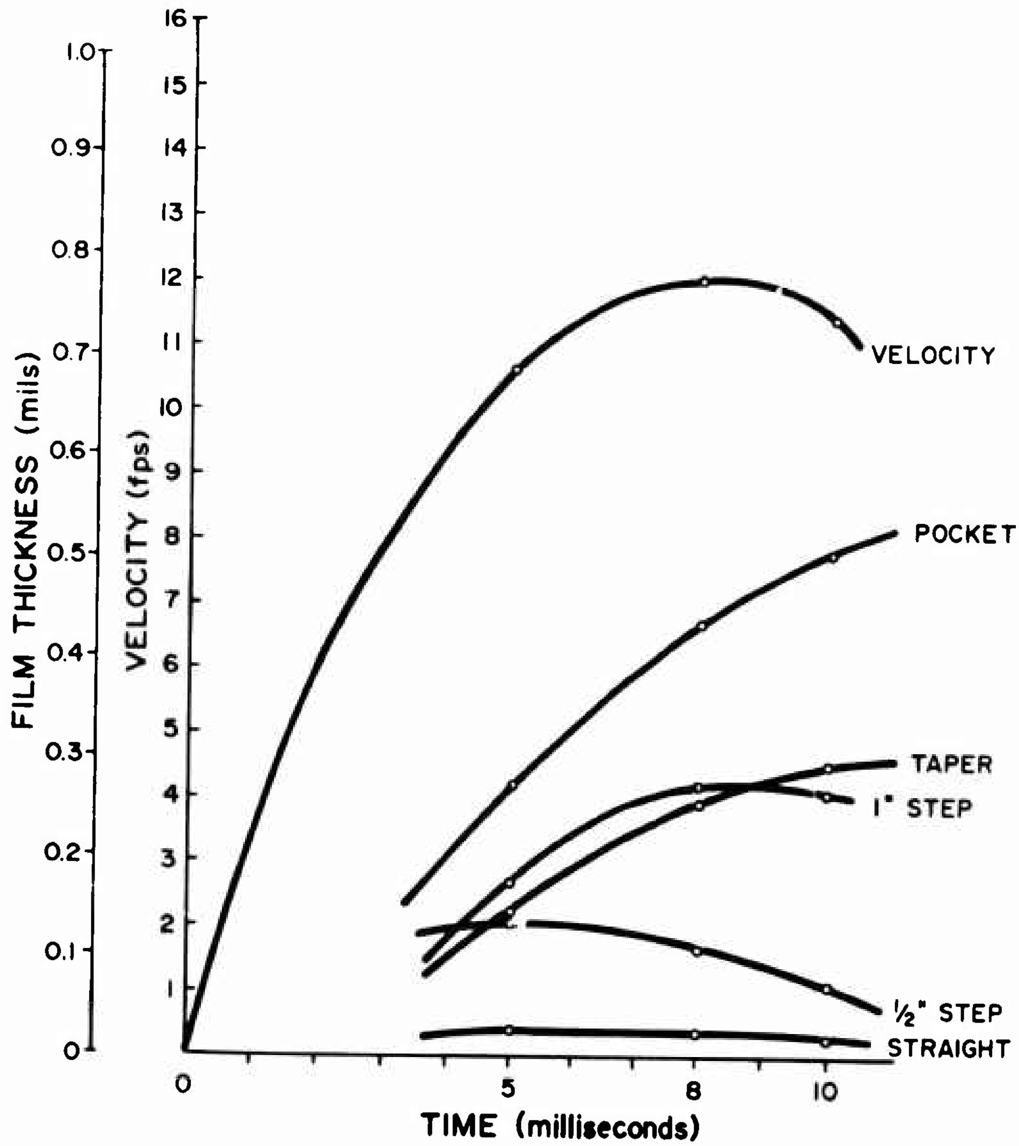

Fig. 8 Ave--age oil film thicknesses for the various bearing surface profiles as functions of time for 1055 psi bearing pressure. The test recoil velocity at this particular bearing pressure is included on the same graph.

TABLE I - Average Oil Film Thicknesses as Functions of Bearing Pressure and Time for the Various Surface Profiles

		Straight surface Profile			Surface profile with taper			Rayleigh Step at 1/2 inch			Rayleigh Step at 1 inch			Surface profile with pocket		
Press (psi)	Time (ms)	Av. Thick. (mils)	No. Meas.	Std. Dev. (mils)	Av. Thick. (mils)	No. Meas.	Std. Dev. (mils)	Av. Thick. (mils)	No. Meas.	Std. Dev. (mils)	Av. Thick. (mils)	No. Meas.	Std Dev. (mils)	Av. Thick. (mils)	No. Meas.	Std Dev (mils)
476	5	1.4	86	1.19	3.5	74	1.38	1.1	74	0.94	5.6	77	0.86	9.1	71	2.49
	8	2.4	94	1.84	7.8	75	1.59	1.8	74	2.20	8.8	77	1.47	16.1	68	2.86
	10	2.6	86	2.04	10.4	75	2.00	1.7	74	2.62	9.7	77	1.67	19.2	65	3.37
772	5	0.1	60	0.39	3.3	79	1.15	2.7	64	0.98	3.8	54	0.76	5.6	117	1.85
	8	0.3	63	0.53	6.6	78	2.19	3.0	64	1.84	5.5	54	1.36	9.5	117	2.68
	10	0.3	63	0.58	8.1	77	2.60	2.5	64	2.07	5.2	54	1.73	11.1	111	3.22
1055	5	0.4	72	0.59	2.2	92	0.65	2.1	69	1.09	2.7	72	0.61	4.2	72	1.06
	8	0.4	71	0.80	3.9	86	1.15	1.7	69	1.48	4.2	72	1.01	6.7	68	1.56
	10	0.3	71	0.73	4.5	84	1.38	1.1	69	1.39	4.1	72	1.40	7.9	63	1.70

from the leading edge showed a thicker film at the intermediate bearing pressure can probably be attributed to uncertainty in the measurements with these thin oil films rather than to an actual optimum.

Pocket bearing

The blind pocket in the loaded sector of the bearing was the best surface profile of those studied. Although the pocket was less than a mil deep, it resulted in a faster acting bearing and an appreciably thicker oil film. At the highest bearing pressure, the 10 millisecond film thickness was almost twice that of the next best surface profile studied.

CONCLUSION

An object of the design of a concentric recoil bearing is to insure reliable full-film lubrication over as much of the recoil and counter recoil motions as possible. The counter recoil is much less critical because both bearing loads and sliding velocities are very much lower. The surface profile has a great effect on the oil film produced in a recoil bearing. A straight surface profile results in exceedingly thin oil films in heavily loaded bearings. Furthermore, the bearing is unreliable and frequently a supporting oil film is not formed at all. The use of a Rayleigh step results in a thicker oil film but it is still not of satisfactory thickness and it collapses rapidly when the velocity decreases probably owing to excessive side leakage. This could lead to increased wear and "hanging-out-of-battery". A small taper on the leading edge results in a good oil

film but the best bearing surface profile of those tested was clearly the one with a shallow blind pocket in the loaded sector. The optimum proportions of the pocket (probably pockets for a large concentric recoil bearing) were not determined and would, in any case probably be different for different bearings but this general design holds great promise.

ACKNOWLEDGEMENT

The recoil simulator was constructed by C. A. Albrecht.

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1. Lord Rayleigh, "Notes on the theory of lubrication", Phil. Mag. 35, (1918)

2. A. Cameron, "Principles of Lubrication", p. 150-151, John Wiley and Sons Inc., (1966)